LOCAL HEAT TRANSFER COEFFICIENT AND PRESSURE DROP OF TWO PHASE STEAM IN A VERTICAL TUBE

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EDWARD B. MORRISON

LOCAL HEAT TRANSFER COEFFICIENT AND PRESSURE DROP OF TWO PHASE STEAM IN A VERTICAL TUBE

by

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Lieutenant, United States Navy

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PREFACE

In the early days of steam generation for power, saturated steam was used a great deal, but in order to achieve higher thermal efficiencies and reduce errosion of steam turbine blading the steam power industry rapidly extended steam conditions into the superheat region. Except for a small amount of data recorded for boiling in vertical tubes with free convection no extensive study had been made of the heat transfer and pressure drop characteristics of wet steam.

With the advent of the nuclear reactor (a highly concentrated heat source) and the forced circulation boiler, wet steam has again become an important energy transfer medium. Basic data for wet steam under forced circulation has become very important. This topic has been investigated in limited ranges by several students at the U. S. Naval Postgraduate School: in 1953 by Fisher and King (3); in 1954 by Davis and Duacsek (1); and in 1955 by Nelson (12). Dengler (2) measured heat transfer coefficients and pressure drop of wet steam at low pressure in a vertical tube at the Massachusettes Institute of Technology in 1953.

The primary purpose of this investigation was to cover the range from 0 to 100% moisture, with particular attention to the annular flow region; to measure heat transfer and pressure drop characteristics and to compare the pressure drop data with the methods of prediction developed by Martinelli et al. at the University of California (7), (9), and (10).

The experimental work of this thesis was carried out at the United States Naval Postgraduate School from January to March 1956.

The writer wishes to express his appreciation for the guidance and

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assistance of Professor E. E. Drucker in conducting experimental work and writing this paper, and to Chief F. H. Miechen, USN for his aid in making changes to the experimental set up and for boiler and condenser operation during the tests.

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TABLE OF SYMBOLS AND ABBREVIATIONS

X_{tt}	Martinelli's dimensionless parameter $X_{tt} = \left(\frac{W_\ell}{W_g}\right)^{0.9} \left(\frac{\rho_g}{\rho_\ell}\right)^{0.5} \left(\frac{\mu_\ell}{\mu_g}\right)^{0.1}$
W	Weight flow, (lbs/hr)
ρ	Density, (lbs/ft^3)
μ	Viscosity, (lbs/hr-ft)
ℓ	Subscript to denote liquid
g	Subscript to denote vapor
q	Rate of heat transfer, (BTU/hr)
A	Area, (ft^2)
h	Film heat transfer coefficient, (BTU/hr-ft^2-$^\circ$F)
t_s	Surface temperature, ($^\circ$F)
t	Fluid temperature, ($^\circ$F)
k	Coefficient of thermal conductivity, (BTU/hr-ft-$^\circ$F)
L	Length, (ft)
t_1	Temperature at thermocouple location, ($^\circ$F)
t_2	Temperature at surface, ($^\circ$F)
r_o	Radius to location of thermocouple, (inches)
r_i	Inside radius, (0.5 inches)
D	Inside diameter, (ft)
Re	Reynold's Number, $\left(\frac{\dot{m} D}{A \mu}\right)$
$\dot{m}$	Mass rate of flow, (lbs/hr)
Pr	Prandtl's Number, $(\mu c_p/k)$
c_p	Specific heat, (BTU/lb-$^\circ$F)

h_L	Film heat transfer coefficient for liquid alone, (BTU/hr-ft^2-$^\circ$F)
R_L	Volume fraction of liquid
D_e	Equivalent diameter (See Chapter V)
x	Quality, (Weight percent vapor)
$(1-x)$	Percent moisture
$(\Delta P/L)_{TPF}$	Frictional pressure drop per unit length for two-phase flow, (psi/ft)
$(\Delta P/L)$	Total pressure drop per unit length, (psi/ft)
$(\text{"}\Delta z\text{"}/L)$	Pressure drop due to change in elevation, (psi/ft)
$(\Delta P/L)_\ell$	Frictional pressure drop for an equivalent mass flow of liquid alone, (psi/ft)
ϕ_{1tt}	Martinelli's dimensionless parameter for correlation of pressure drop. $\phi_{1tt}^2 = \dfrac{(\Delta P/L)_{TPF}}{(\Delta P/L)_\ell}$
P_{center}	Pressure at the center of the test section
P_1	Pressure at the test section inlet
Δt_{film}	Surface temperature minus fluid temperature
Q/A	Heat flux, (BTU/hr-ft^2)
ρ_{200}	Density of saturated steam at 200 psig.

CHAPTER I

INTRODUCTION

The problems of two phase flow have long been apparent in special industrial applications, but only specific problems have been investigated leaving a dearth of general data. The characteristics of two phase flow that are of interest in this investigation are the local film heat transfer coefficient and the pressure drop per unit length.

Specific problems of two phase flow have been carried out in the refrigeration and in the chemical process industry. The observed values of film heat transfer coefficient are much greater with two phase flow than with the liquid alone or the vapor alone. Verschoor and Stemerding (13) observed film heat transfer coefficients for two phase flow up to seven times that observed with the liquid alone, in an investigation with air and water. Dengler (2) observed values up to fifteen times greater with wet steam at low pressure.

The primary purpose of this investigation was to determine the values of film heat transfer coefficient over a wide range of moisture contents. In a similar investigation of Freon at low temperature Yoder and Dodge (14) observed a maximum of the film heat transfer coefficient at about 40% dry Freon by weight.

A secondary purpose of this investigation was to measure the pressure drop over a wide range of moisture contents and compare the results with the prediction methods of Martinelli, et al. (7), (9), and (11). Generally the experiments of Martinelli were carried out isothermally in horizontal pipes.

The primary variables in this investigation are: the mass rate of flow of the liquid, the velocity of the liquid, the viscosity of the liquid, the density of the liquid, the mass rate of flow of vapor, the velocity of the vapor, the viscosity of the vapor, the density of the vapor, and the rate of heat transfered per unit area. In two phase flow with the equipment available the velocity of the liquid or the vapor could not be determined. With annular flow the vapor flows through the center of the pipe at high velocity while the liquid flows relatively slowly along the sides. Measurements of the volumetric fraction of liquid for two phase flow in horizontal pipes have been made by Martinelli (7), (9), and (10) by operating two quick closing valves to isolate the section, followed by a rapid blow down to remove the vapor. The liquid was then drained and measured. To measure the water clinging to the walls of the test section a volatile fluid was used to rinse the section. This was collected; the volatile liquid was distilled off, and the remaining water was measured. Another complicated method was used by Dengler (2) to measure the volumetric fraction in a vertical test section. Radioactive tracer was added in minute quantity to the liquid. A Geiger-Mueller counter was moved up and down outside of the test-section, and the resultant counting-rate measurement, together with the local weight fraction vapor and the liquid density, could be related to obtain the volumetric fraction. The above work of Martinelli and Dengler make it possible to estimate the volumetric fraction for this experiment.

The magnitude of the flow rates in pounds per hour of the liquid and vapor were measured by flow meters after separation by a centrifix. The viscosity and density were determined from the steam tables as a

function of saturated pressure. To determine the heat transfer coefficient: thermocouples were used to determine wall temperature; and the saturation temperature corresponding to the local pressure was used for fluid temperature. In this investigation pressure was held constant except for three runs. Therefore, viscosity and density of the liquid and the vapor were constant. Isothermal runs were made to determine pressure drop at 400, 700, and 1000 pounds per hour. Heated runs were made with one heat rate (78,300 BTU/Hr/Ft^2) at 400, 700, 1000, and 1250 pounds per hour. The limitations of the equipment were investigated and these are included in Appendix II.

The vertical test section included four separately heated six inch sections in series with an inside diameter of $\frac{1}{2}$ inch. The pressure drop was determined from the value measured across the two center sections. The heat transfer coefficient was determined as the average heat transfer coefficient for the two center test sections.

The results of this experiment were correlated with percent moisture as well as with X_{tt} (a dimensionless parameter developed by Martinelli (7)). Heat transfer coefficients and pressure drops are usually related in single phase investigations to several dimensionless quantities, such as Nusselt, Reynolds, and Prandtl numbers. Such correlations for single phase flow are not adequate because these numbers are different for each phase and do not take into account the configuration of the liquid-vapor flow. The dimensionless X_{tt} is used in an attempt to correlate the known properties of each component with pressure drop and heat transfer characteristics.

CHAPTER II

TWO PHASE FLOW

A number of studies have been made of the complex configurations of two phase flow. These studies invariably are broken down to take into account the differences in the flow configuration. A designation is given to each type of flow. The flow configuration actually varies in many ways but several combinations of the simple designations will suffice to discribe the flow pattern in this investigation. These basic configurations are:

(a) clear ----- dry vapor,

(b) mist ------ tiny water droplets carried along in the main vapor flow at low moisture contents,

(c) annular flow -- characterized by the fact that the liquid flows in an annulus along the tube wall, while the vapor passes at a much higher velocity through the center of the tube,

(d) slug flow -- alternate slugs of liquid and vapor pass through the tube,

(e) bubble flow -- small vapor bubbles pass individually through the tube at low vapor contents,

(f) pure liquid flow -- no bubbles.

These types of flow were observed from top to bottom as the percent moisture was increased from zero to 100%. The transition between two types of flow was an over lapping affair and depended on the flow rate as well as the moisture content (slug flow occured at lower percentage of moisture at the lower flow rates). For example at high moisture

contents with annular flow there would invariably pass an occasional slug with rather consistent regularity, and at low moisture contents when annular flow was starting to occur there was certainly still a definite mist in the center of the tube. The description of the flow configuration definitely depends on the observer. The writer chose to keep the description simple and only of secondary nature in this investigation. The flow designations above were first described by Bergelin (15) for flow through a vertical tube. A considerably different configuration was described by Martinelli (9) for flow in a horizontal tube. The basic difference for horizontal flow is that the liquid stays on the bottom of the tube while the vapor travels at high velocity through the top portion. Even so, at certain velocities an assymmetrical type of annular flow was noted in the horizontal tube. This configuration difference accounts for the differences in results obtained by Martinelli (8) and Dengler (2).

A survey of the literature on two-phase flow has recently been made by Isbin (4). The two most widely used methods of predicting two-phase pressure drop are the Martinelli correlations and the friction factor methods. Of the two methods, the Martinelli correlations have received the most support and have been chosen for comparison in this investigation. However, Isbin notes several drawbacks to the Martinelli method. The total flow rate parameter is not adequately provided for and the system pressure parameter required further treatment.

A good general discussion of the work of some of the recent developments is included in the latest edition of McAdams (8).

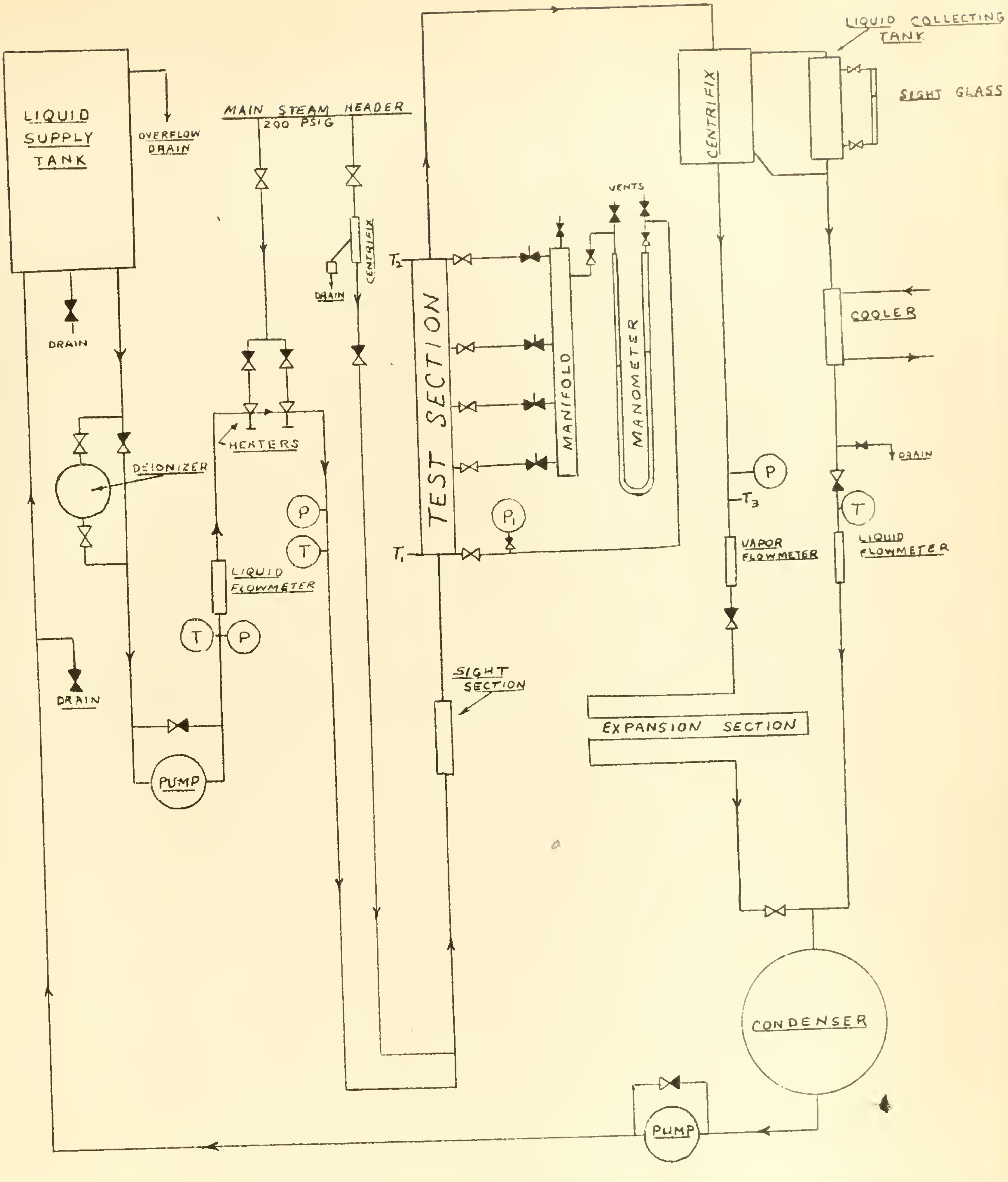
CHAPTER III

EQUIPMENT

The experimental layout presented in the schematic diagram of Figure 1, was obtained by modification of the set up used by Nelson (11).

The equipment consisted essentially of a flow system in which steam was taken from the main steam line, through a gate valve, a small centrifix seperator, and a throttling valve, passed vertically downward through 15 feet of $1\frac{1}{2}$ inch piping and then vertically upward for 8 feet ($1\frac{1}{2}$ inch pipe), through three feet of flexible hose ($\frac{1}{2}$ inch), a sight section ($\frac{1}{2}$ inch), another three feet of flexible hose to the test section. The purpose of the flexible hose was to allow for expansion and prevent undue stress on the sight section. Saturated water or wet steam was injected into the stream on the lower end of the upward flow section. Early in the test a nozzle was used to spray the water into the main steam line. Later it was found that a greater range of moisture content due to the increased stability of the system could be obtained without the nozzle, with no apparent change in the flow distribution of the system. The purpose of the loop in the steam line was to allow the steam-water mixture to approach equilibrium in a long straight section before entering the test section.

At the exit of the test section the flow was reversed and passed vertically downward through a Centrifix Type RA Separator. From the steam outlet of the centrifix the dry steam was passed through a Fisher and Porter Co., Series 50 Flowrator meter, then through a needle control



SYMBOLS

⧓ QUICK ACTING TOGGLE VALVE

⋈ NORMALLY OPEN VALVE

⧓ NORMALLY CLOSED VALVE

⧑ THROTTLED

T_1, T_2, T_3 - STREAM THERMOCOUPLES

Ⓣ - THERMOMETERS

E. B. Morrison

TEST SET-UP (SCHEMATIC) FIGURE 1

valve. (This valve was changed from preceding the steam flowmeter as was used in Nelson's investigation (11), to after the flowmeter in order to insure dry saturated vapor in the flowmeter.). After the steam exit control valve, the steam passed through 16 feet of 3/4 inch pipe, 35 feet of 1½ inch pipe, seven feet of two inch pipe and discharged into a condenser operated at a slight vacuum. From the water outlet of the centrifix the saturated water passed through a cooling coil, a needle valve, a Fisher and Porter Series 700 Flowrator meter, and discharged directly to the condenser. The needle valve was adjusted until the water level of the centrifix remained constant before noting a reading on the flowmeter.

The moisture content of the steam leaving the Centrifix was determined by a U-Path Steam Calorimeter. The moisture content remained below ½% except at high inlet moisture contents with high flow, where the steam leaving the seperator approached one percent moisture.

Steam supply came from a Babcock and Wilcox F M Boiler at an operating preasure of 200 psig.

Condenser condensate was pumped to a supply tank for a source of cold pure water. The water was partly bypassed through a Duolite deionizer on the way to a centrifugal vane type pump, then through a Fisher and Porter Series 700 Flowrator meter, through a needle control valve to a heating section. The heating section consisted of two Schutte and Koerting Co. jet type heaters mounted in series. A separate line was run from the main steam header to supply heating steam. The heaters were more than adequate for all conditions of these tests. From the heater section the saturated water (or under most conditions

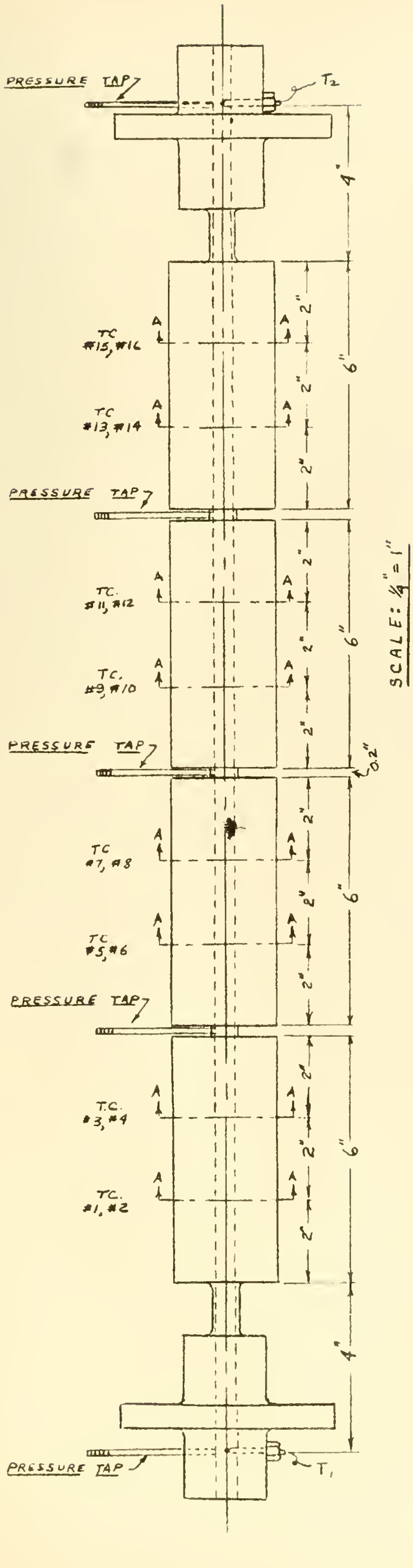
wet steam) passed down to the lower end of the upper flow section where it was mixed with the main steam flow as mentioned earlier in this chapter.

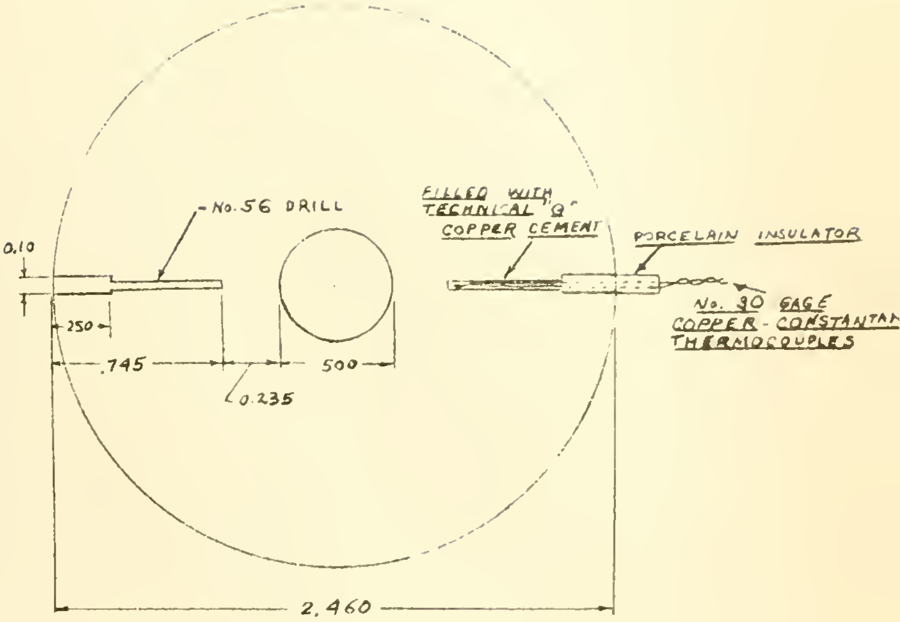
The test section used by Nelson (11) was reworked. All new thermocouples and fittings were installed. No. 30 gage copper-constantan thermocouples were installed as indicated in Figure 2. A Conax Thermocouple gland seal was used to seal the stream thermocouples. Pressure taps were installed at the entrance and exit of the test section and at three places in the test section as indicated in Figure 2.

Heat supply to the section consisted of four independent heating elements containing about 70 feet of No. 17 Nichrome V wire, wound around each of the four sections in a single layer. Each heating element completely covered the section. A thin layer of mica helped insulate the section electrically from the heating wire. The power supply to the heaters was controlled by four two -gang variac assemblies and measured by portable wattmeters.

Heat insulation of the test section was accomplished by wrapping several layers of glass tape on the outside of the heating coils, covering with a thick layer of asbestos cord, and over this was fitted a $2\frac{1}{2}$ inch layer of magnesia brick, and a $\frac{1}{2}$ inch of wet magnesia mix. Cheesecloth was wrapped over the insulation and held with cornstarch paste to prevent flaking off. The centrifix and all piping up to the steam flowmeter was also insulated. Heat loss was determined to be negligible through the insulation.

A differential manometer was used to measure pressure drop in the section. The inlet pressure tap was connected to one side of the manometer as a reference. A pressure gage was also connected to this tap




No. 56 DRILL
FILLED WITH TECHNICAL "Q" COPPER CEMENT
PORCELAIN INSULATOR
No. 30 GAGE COPPER-CONSTANTAN THERMOCOUPLES
0.10
.250
.745
0.235
.500
2.460

SECTION AA
SCALE: 1" = 1"

THERMOCOUPLE DETAIL

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Material: Copper-Tellurium
(99.35% Cu, 0.65% Te).

Conductivity: 222 BTU/hr-ft-°F

FIGURE 2

to measure inlet pressure. To the other side of the manometer was connected a manifold to which the other four pressure taps were connected. The manometer lines were connected at the test section by Norgren needle valves and at the manifold by quick acting toggle valves manufactured by Hoke. During the experiment the pressure tap lines were kept full of water to insure a known water level. To keep these lines full of water, wet rags were applied to the lines to condense any steam entering.

It has been determined as a result of this experiment that it would have been better to use the center pressure tap as the reference pressure. Because of end leakage of heat, only the second and third sections were used for heat transfer calculations. At high flow rates the pressure drop from the inlet to the center is significant. By moving the inlet pressure gage to the center of the test section, the average test section pressure can more easily be held constant.

All components of the system were constructed, as far as practicable, of non-ferrous material.

CHAPTER IV

OPERATING PROCEDURE

At the begining of each operating period the test section was filled with cold water from the supply tank and the manometer system was filled with water by bleeding off any bubbles in the line accumulated since the previous operating period. Then the system was flushed by flowing a large quantity of steam through the test section to the condenser. The condensate was allowed to run down the drain until it was clear and free of particles. The drain was then secured and the condensate by-passed to the storage tank. This procedure required about two hours.

The operating variables that could be controlled were pressure, heat flux, flow rate, and moisture content. As noted in Appendix II certain limitations of flow existed for each pressure held at the test section. The inlet pressure to the test section was maintained at 104 psig for all runs except three runs at 149 psig. The heat input per section was limited to 1.5 KW for all runs. At one KW per section the experimental error was magnified. At two KW per section the circuit drew excessive current.

A total of 69 runs were made of which 50 were considered to be satisfactory. A summary of the data is listed in Appendix I. Unsatisfactory runs were caused by instability of the system making it difficult to maintain equilibrium conditions at a desired pressure. The unsatisfactory runs are not included in the appendix.

The desired flow rate and moisture content was principally obtained

equilibrium. For the heated runs the heaters were turned on to ½ KW per section after the liquid flow had been established. After a period of time, during which steam adjustments were made, the heaters were increased to one KW per section and then 1½ KW per section. If the increase was too rapid, excessive current caused the fuses to blow. The time required to adjust to a desired point varied from 30 minutes for an isothermal run up to an hour or more for a heated run. The stability of the system greatly affected the time required to adjust to a desired value. The system approached instability at low flow rates and at high moisture content runs for all flows as the system approached slug flow. The stability was greatly improved over the whole range when the water nozzle was removed. The stability for a desired condition could sometimes be improved by reducing the heating liquid to below saturation and replacing it with steam passing through the main steam line. At other times the main steam line was cut off altogether; the water was heated with one heater, and the pressure was controlled by the throttle valve to the other heater.

For each run, thermocouple millivolt readings, test section inlet pressure, differential pressure drops, water flowmeter reading, steam flowmeter reading, and pressure, and heat input in KW were recorded.

Several additional runs were made to check the validity of the data. The steam flow meter was calibrated. The amount of water contained in two inches of the centrifix was determined for use at very low moisture content, and extensive comparisons of thermocouple readings were made. Pure liquid runs were made to evaluate end leakage and validity of heat transfer calculations. These calibration runs are more thoroughly described in Appendix II.

CHAPTER V

METHOD OF CALCULATION

The film heat thransfer coeficient, h, is defined as the proportionality factor in Newton's Law of Cooling (8):

$$dq = h\ dA\ (t_s - t)$$

The surface temperature (t_s) was determined from the average of the thermocouple readings in the second and third sections, taking into account the temperature difference between the surface and the location of the thermocouples. This temperature difference was determined to be 5.02 °F, for Q/A equal to 78,300 BTU/hr-ft^2 (corresponds to 1.5 KW per section), from the Fourier conduction equation for a circular tube (8):

$$q = \frac{2\pi k\ L\ (t_1 - t_2)}{\ln\ r_o/r_i}$$

The temperature of the fluid (t) in Newton's equation was taken as the saturation temperature corresponding to the pressure at the center of the test section. Because of thermal entrance effects on the first section and end leakage from the first and last sections, only the two center sections were used in the final calculations.

The pressure at the center of the test section was determined from the inlet pressure gage reading less the pressure drop indicated by the differential manometer.

The moisture content was taken at the exit of the test section. The amount of liquid evaporated (about 20 lbs/hr) was very small in proportion to the total flow.

The single phase points, pure liquid and pure vapor, were calculated from the convection equation for flow in tubes (8):

$$(hD/k) = 0.023\,(Re)^{.8}(Pr)^{.4}$$

The calculations for the liquid phase (h_L) were used for the comparison with Dengler's results (h/h_L vs $1/X_{tt}$), Figures 4 and 5.

The pressure drop was measured by a differential manometer across the second and third sections. The single phase pressure drops were calculated from standard friction factor charts.

The dimensionless parameter X_{tt} was developed from dimensional analysis by Martinelli (9):

$$X_{tt} = \left(\frac{W_l}{W_g}\right)^{0.9} \left(\frac{\rho_g}{\rho_l}\right)^{0.5} \left(\frac{\mu_l}{\mu_g}\right)^{0.1}$$

It has been found useful in correlating data of two phase flow over a wide variety of conditions. X_{tt} was calculated as a function of x for two different pressures (104 psig and 149 psig) with data from the steam tables for correlation of the data of this investigation.

It appeared to be possible to predict values of h/h_L from measured values of R_L, or visa versa, with the assumptions that there is annular flow, that there is no nucleate boiling, and that an equivalent pipe size can reqresent the cross section of the liquid area. An equivalent diameter D_e is defined to represent the equivalent pipe. For single phase forced convection: (using equivalent pipe).

$$hD_e/k = 0.023\,(Re_e)^{.8}\,(Pr)^{.4}$$

where $Re_e = D_e\,\dot{m}\,(1-x)/A\mu$,

or to obtain h/h_L rearrange and divide by h_L in the same form:

$$h/h_L = \frac{0.023\,(k/D_e)\,(Re_e)^{.8}\,(Pr)^{.4}}{0.023\,(k/D)\,(Re)^{.8}\,(Pr)^{.4}}$$

Simplifying,

$$h/h_L = \left[\frac{D^{2.25}(1-x)}{D_e^{2.25}}\right]^{.8}$$

Since $D_e = D\sqrt{R_L}$ by definition,

$$h/h_L = \left[\frac{(1-x)}{R_L^{1.125}}\right]^{.8}$$

or rearranging,

$$R_L = \left[\frac{(1-x)}{(h/h_L)^{1.25}}\right]^{.889}$$

Similar attempts to calculate pressure drop were not successful.

CHAPTER VI

RESULTS AND CONCLUSIONS

The measured heat transfer coefficient vs percent moisture by weight is shown in Figure 3. It is noted that in the annular flow region (about 30% to 80% moisture) the variation is essentially linear. From the initial observation of these results it appears that a relatively unpredictable mechanism is causing an increase in the local film heat transfer coefficient. But the results of these tests combined with the work of Dengler (2) indicate that the local film heat transfer coefficient for vertical two phase flow is still a single phase phenomena in the annular flow region. The only significant resistance to heat flow is the liquid laminar boundary layer. For the heat flux used in this investigation no nucleate type of boiling occured at the metal surface, and the substantial increase in heat transfer coefficient can be attributed entirely to the increase in liquid velocity and the consequent reduction of the laminar boundary layer thickness.

It will also be noted in Figure 3 that the pressure is a very significant variable. Dengler (2) has suggested plotting h/h_L vs $1/X_{tt}$ and this brings together the runs at different flow rates (Figure 4) as well as at different pressures. The pressure, however, has not been completely provided for as is seen by comparing the results (Figure 4) of this investigation at 104 psig with Dengler's results at about atmospheric pressure. Difference in pipe size (Dengler used 1" diameter compared to ½" diameter for this investigation) may partly cause the difference in results.

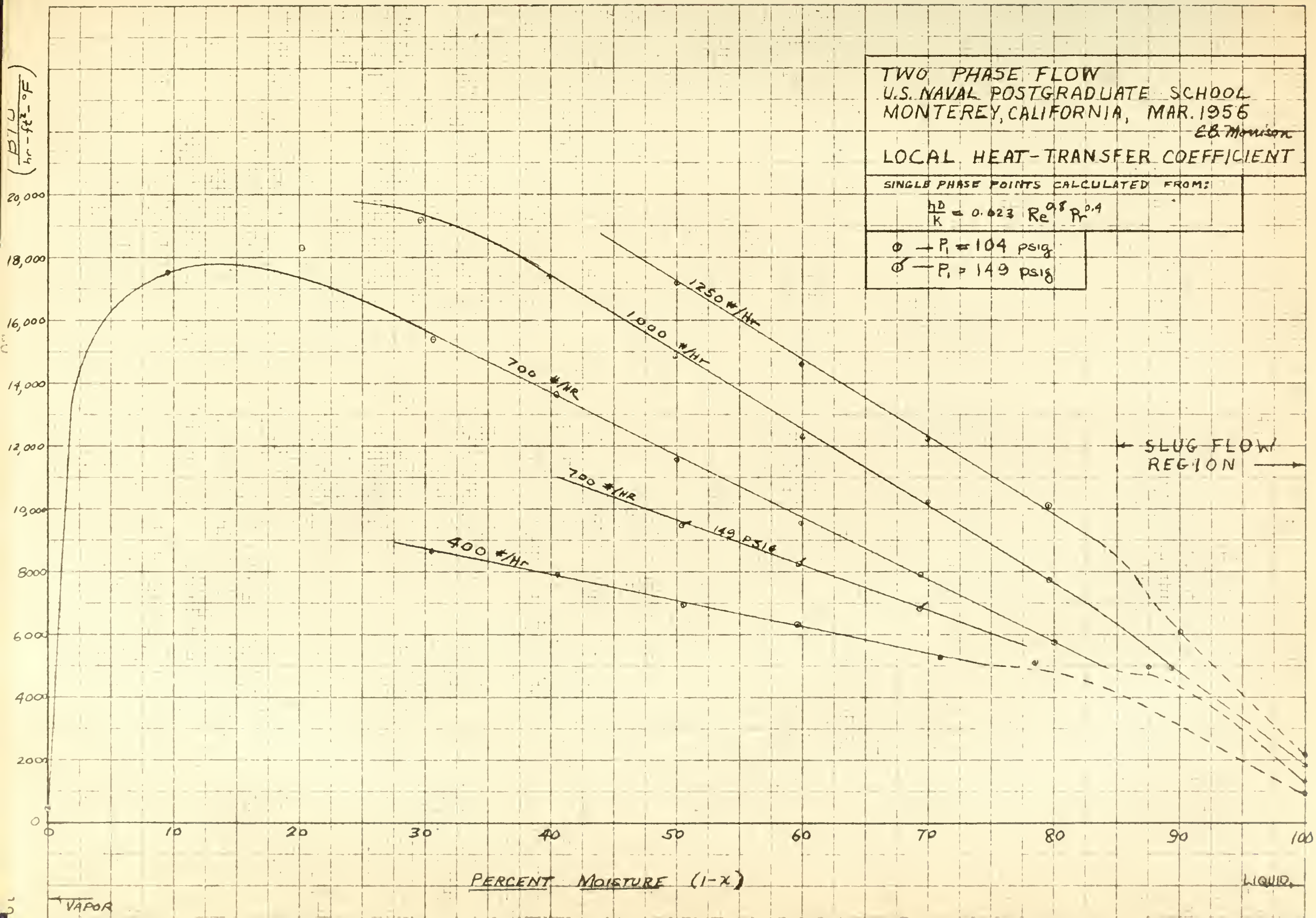
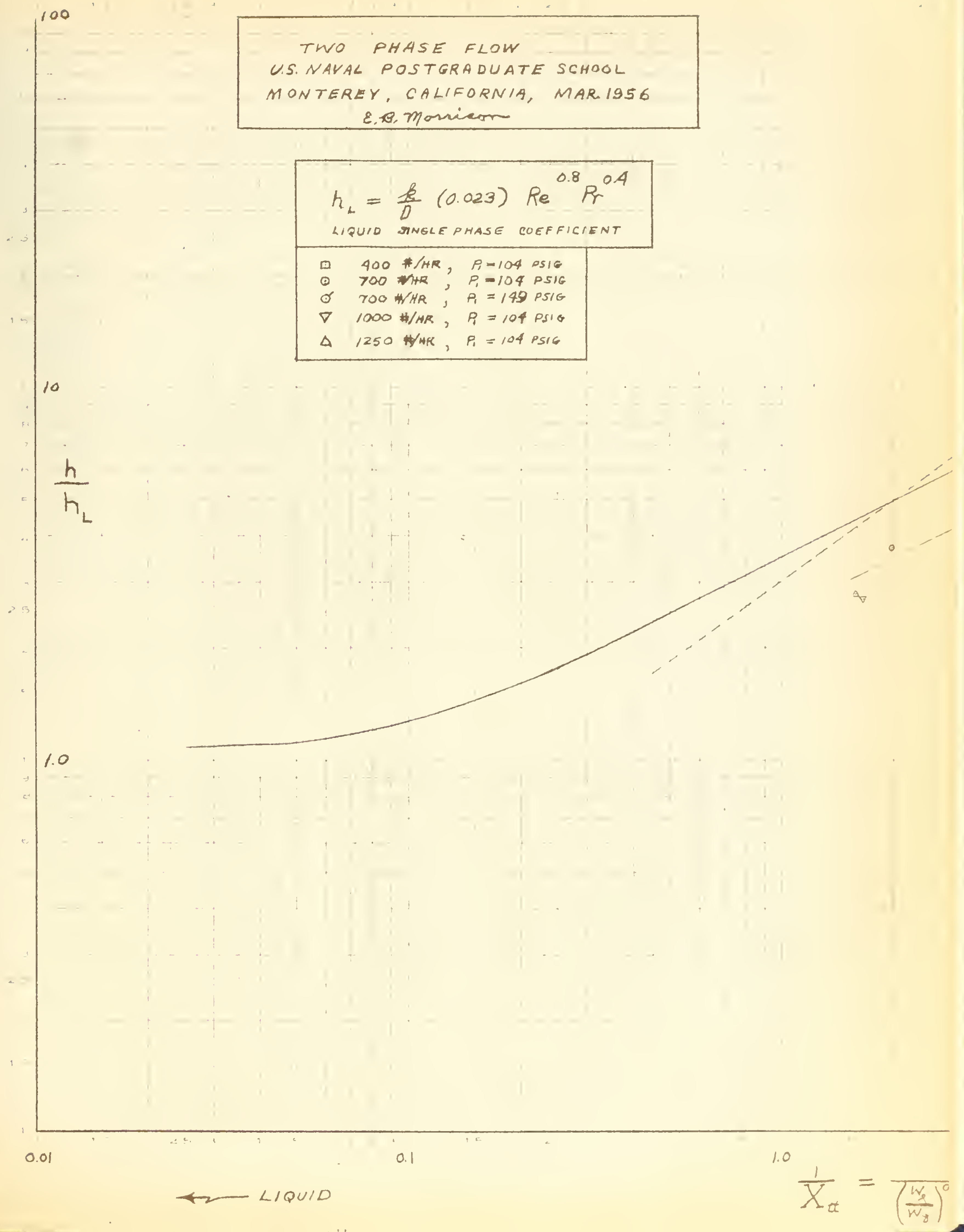
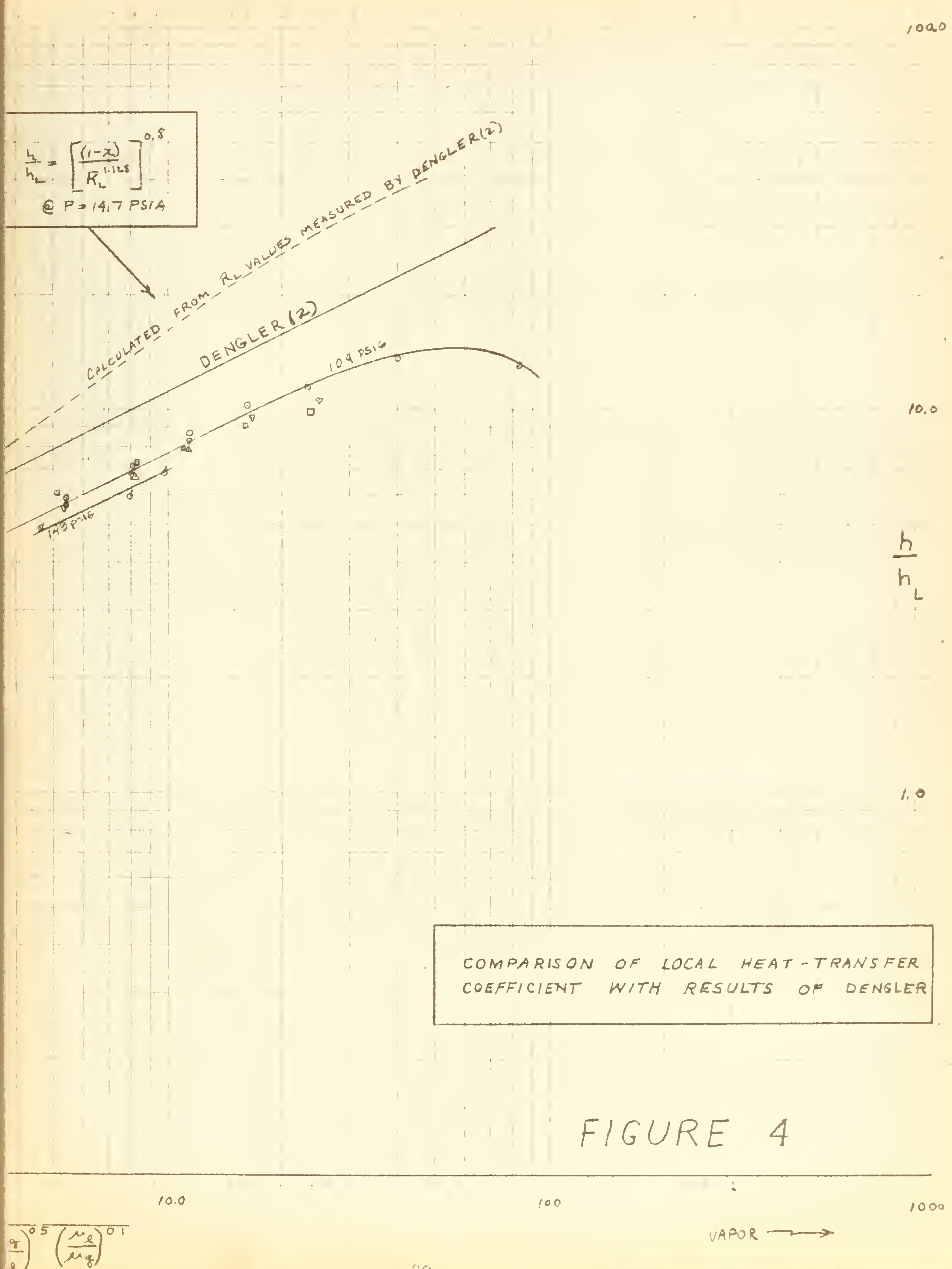

TWO PHASE FLOW
U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIFORNIA, MAR. 1956
E.B. Morrison
LOCAL HEAT-TRANSFER COEFFICIENT
SINGLE PHASE POINTS CALCULATED FROM:
$\frac{hD}{K} = 0.023\ Re^{0.8}\ Pr^{0.4}$
ϕ — P_1 = 104 psig
ϕ — P_1 = 149 psig
1250 #/Hr
1000 #/Hr
700 #/HR
700 #/HR 149 PSIG
400 #/Hr
← SLUG FLOW REGION →
$\left(\frac{BTU}{hr - ft^2 - {}^\circ F}\right)$
20,000
18,000
16,000
14,000
12,000
10,000
8000
6000
4000
2000
0
0
10
20
30
40
50
60
70
80
90
100
PERCENT MOISTURE (1-x)
← VAPOR
LIQUID

FIGURE 3


TWO PHASE FLOW
U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIFORNIA, MAR. 1956
E. B. Morrison
$h_L = \frac{k}{D}(0.023)\ Re^{0.8}\ Pr^{0.4}$
LIQUID SINGLE PHASE COEFFICIENT
□ 400 #/HR, P_1 = 104 PSIG
⊙ 700 #/HR, P_1 = 104 PSIG
⊙ 700 #/HR, P_1 = 149 PSIG
▽ 1000 #/HR, P_1 = 104 PSIG
△ 1250 #/HR, P_1 = 104 PSIG
100
10
1.0
$\frac{h}{h_L}$
0.01
0.1
1.0
← LIQUID
$\frac{1}{X_{tt}} = \frac{\ }{\left(\frac{W_l}{W_g}\right)^{0}}$



COMPARISON OF LOCAL HEAT-TRANSFER COEFFICIENT WITH RESULTS OF DENGLER

FIGURE 4

Dengler (2) also measured (R_L) the volumetric fraction of liquid (See Chapter I). Using Dengler's values of R_L (Figure 5) and a pressure of 14.7 psia, h/h_L was calculated from single phase considerations (See Chapter IV for method of calculation) and plotted on Figure 4. The deviation can be attributed mainly to the degree to which an equivalent pipe does not represent the actual flow at the surface of the tube. In Dengler's experiments the pressure varied from 7 psia to 29 psia and this may also cause some of the discrepancy. Even with the discrepancy as shown it is felt that the hypothesis that the film heat transfer coefficient is a single phase problem is substantiated.

To carry the comparison one step further, R_L was calculated from the experimental points (h/h_L) of this investigation and plotted on Figure 5. A definite shift upward is noted for the increase in pressure This trend is also noted by Martinelli (11) for horizontal flow, but only in a qualitative manner. This investigation suggests that the shift in R_L (also h/h_L) is much greater in the low pressure regions than at higher pressures. This further suggests that a modification of the density term in X_{tt} is required to give a better correlation.

The measured values of pressure drop per unit length vs percent moisture are shown in Figure 6. This pressure drop is composed of three terms:

1. The frictional two phase pressure drop, $(\Delta P/L)_{TPF}$.
2. The pressure drop due to change in elevation, "Δz"/L, (psi/ft).
3. The pressure drop due to change of momentum during evaporation. This value was negligible in this investigation because of the low vaporization. This fact is verified by the indistinguishable difference between the heated and the isothermal points (Figure 6).

TWO PHASE FLOW
U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIFORNIA, MAR. 1956
E B Morrison

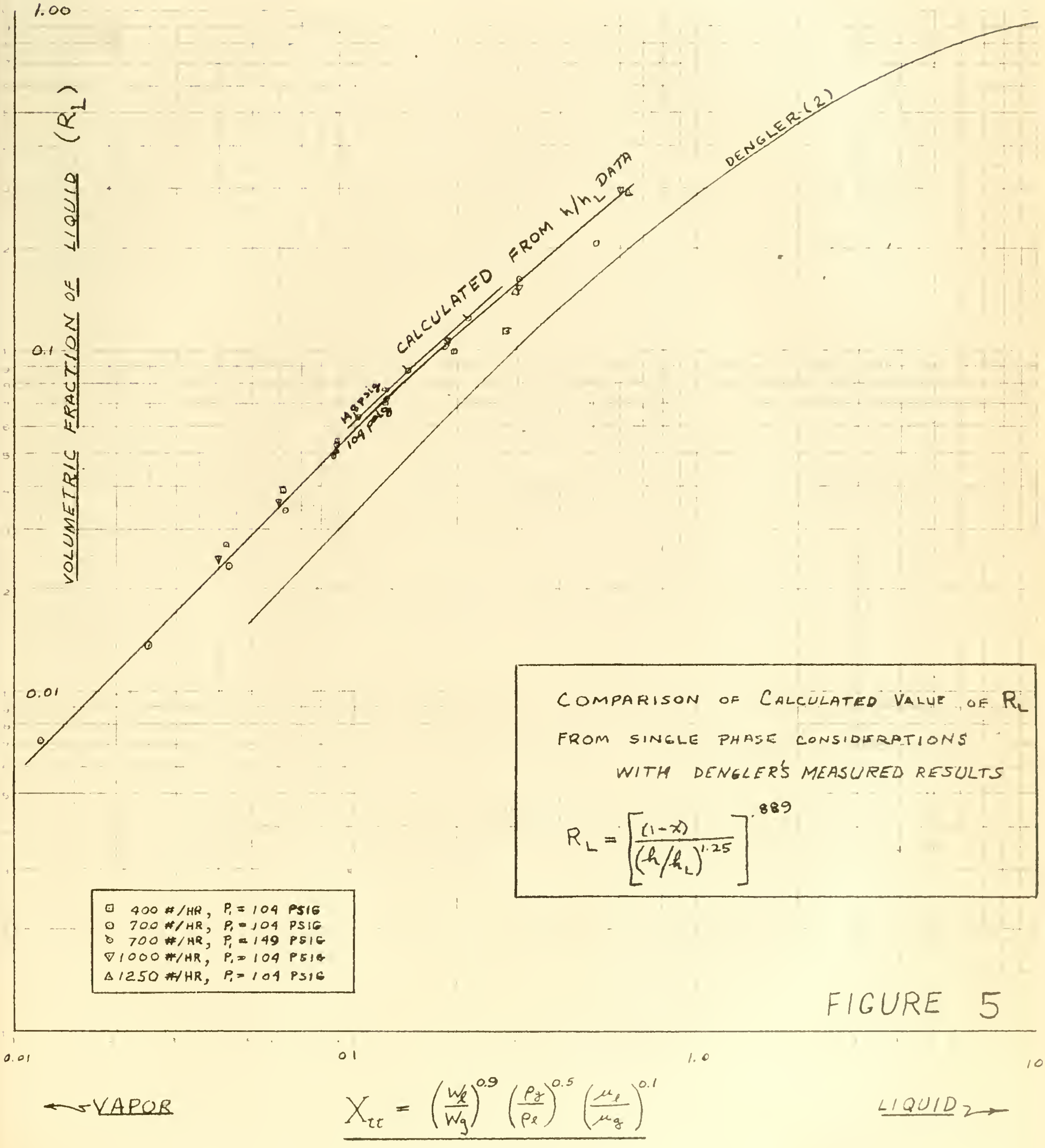


FIGURE 5

For correlation of pressure drop Martinelli (7) and (11) has suggested the use of a parameter $\phi_{1_{tt}}$ where: $\phi^2_{1_{tt}} = \frac{(\Delta P/L)_{TPF}}{(\Delta P/L)_l}$ . .

The numerator represents the frictional component of two phase flow and the denominator represents the frictional pressure drop for an equal mass rate of liquid alone. In Figure 7, $\phi^2_{1_{tt}}$ vs X_{tt} is plotted for this investigation and compared with the results of Dengler. The dashed line indicates the possible range of Dengler's curve at the left end. The difference of results is more pronounced in this comparison than is easily explained by the difference in operating pressure. Martinelli (11) again shows a trend in this direction with increasing pressure but not to this extent. It is believed by the writer that the pipe size becomes more important for the pressure drop correlations. This added to the difference in operating pressure may account for the large discrepancy.

Initial attempts to calculate frictional pressure drop from single phase considerations (knowing R_L) were unsuccessful. It is believed that again pipe size is the significant variable that was not adequately taken into account.

Summary:

1. The film heat transfer coefficient is linear in the annular flow region when plotted against percent moisture.
2. The only significant resistance to heat flow is the liquid laminar boundary layer, and hence the increase in h is due primarily to liquid velocity increase.
3. The ratio h/h_L vs $1/X_{tt}$ correlates well and compares favorably

with results of Dengler (2).

4. The ratio h/h_L can be calculated approximately from R_L data (in the annular flow region) by modifying single phase methods.

5. Effects of pressure on h/h_L are much greater for the low pressure regions.

6. The pressure drop parameter $\phi^2_{l_{tt}}$ correlates well with X_{tt}.

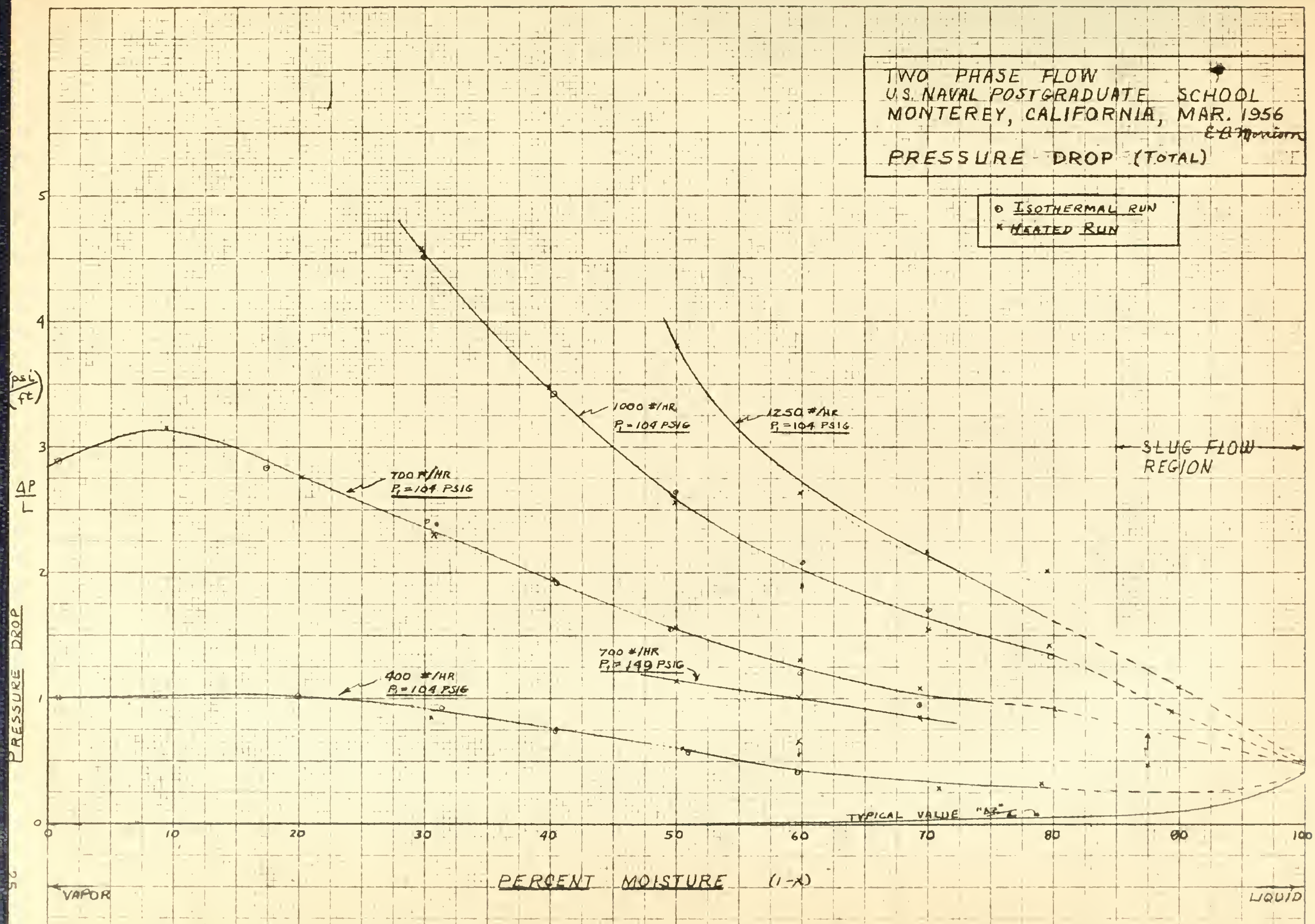

TWO PHASE FLOW
U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIFORNIA, MAR. 1956
E.B. Morton
PRESSURE DROP (TOTAL)
o ISOTHERMAL RUN
x HEATED RUN
1000 #/HR P_1 = 104 PSIG
1250 #/HR P_1 = 104 PSIG
700 #/HR P_1 = 104 PSIG
700 #/HR P_1 = 149 PSIG
400 #/HR P_1 = 104 PSIG
← SLUG FLOW REGION →
TYPICAL VALUE "ΔZ"
PRESSURE DROP $\frac{\Delta P}{L}$ (psi/ft)
0
1
2
3
4
5
PERCENT MOISTURE (1-x)
0
10
20
30
40
50
60
70
80
90
100
← VAPOR
LIQUID →

FIGURE 6

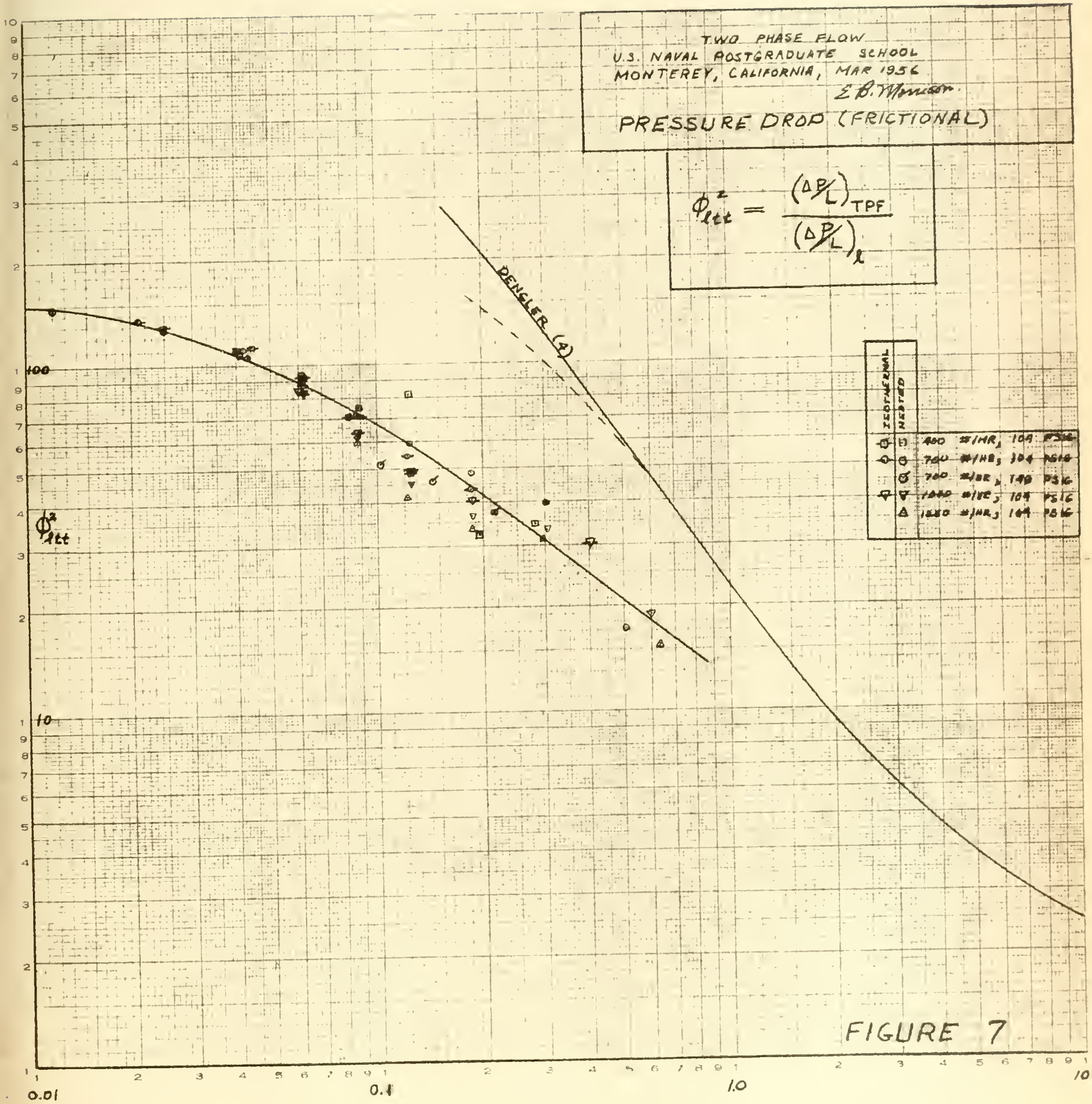


FIGURE 7

$$X_{tt} = \left(\frac{W_l}{W_g}\right)^{0.9} \left(\frac{\rho_g}{\rho_l}\right)^{0.5} \left(\frac{\mu_l}{\mu_g}\right)^{0.1}$$

← VAPOR

LIQUID →

CHAPTER VII

RECOMMENDATIONS

Several significant problems are suggested by the results of this investigation:

1. The prediction of two-phase film heat tranfer coefficients from the direct measurement of the volume fraction of liquid in an isothermal experiment.

2. Conversely, the indirect measurement of the volume fraction of liquid (particularly at higher pressures) from the measurement of the local film heat transfer coefficient. Along with this investigation the effects of smaller test section diameter could be investigated.

3. The analytical prediction of two phase pressure drop from a single phase approach. This would assume that the frictional pressure drop is due to the increase in the liquid velocity only.

Certain changes are suggested in the construction of the test section to avoid some of the difficulties encountered in this investigation. The reference pressure and the inlet pressure gage should be moved to the center of the test section since this is the significant pressure in the ultimate results. The inlet and outlet pressure taps should be moved from the vicinity of the stream thermocouples to obtain consistent readings of pressure differences along the test section. It is recommended that these pressure taps be installed immediately before and after the heated sections giving equal spacing between all pressure taps. Difficulty was encountered with leakage at the pressure

tap connections. This was caused at the point where the hard brass fitting into the soft test section. The thread could be stripped by installing the fitting only finger tight. It is recommended that more threads and a slightly larger diameter of fitting be used. It should be remembered when designing the test section that the first and last sections will not give significant results because of end leakage and a thermal entrance effect.

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APPENDIX I

SUMMARIZED DATA

MASS FLOW RATE: 700 lbs/hr.
Inlet Pressure: 104 psig

Run No.	Percent Moisture (1-x)	Mass Rate (#/hr)	P_{center} (psig)	X_{tt}	$\Delta P/L$ (psi/ft)	Δt_{film} ^{o}F	Q/A $BTU/hr/ft^2$	h $\frac{BTU}{hr\text{-}ft^2\text{-}^{o}F}$	Flow Type (Key on next page)
1	0.8	705	100.11	.002	2.88		0		M
2	17.4	694	100.16	.021	2.84		0		MhA
3	30.1	704	100.73	.042	2.42		0		MhA
4	30.9	697	100.79	.043	2.38		0		MhA
5	40.5	703	101.41	.062	1.92		0		A
6	49.5	699	101.90	.083	1.55		0		A
7	59.8	702	102.37	.124	1.21		0		As
8	69.3	702	102.70	.185	0.96		0		Ahs
9	9.5	700	99.75	.012	3.15	4.47	78,300	17,520	MhA
10	20.2	704	100.27	.025	2.76	4.28	"	18,300	MhA
11	30.7	699	100.90	.043	2.30	5.08	"	15,400	MhA
12	40.3	701	101.38	.062	1.94	5.74	"	13,650	A
13	50.0	704	101.89	.087	1.56	6.76	"	11,590	A
14	59.8	702	102.24	.124	1.31	8.21	"	9,540	As
15	69.3	702	102.53	.185	1.09	9.94	"	7,880	Ahs
16	80.0	700	102.75	.303	0.92	13.61	"	5,750	S
17	87.5	701	103.69	.504	0.47	15.70	"	4,980	Sh
Inlet Pressure:		149 psig							
18	50.3	699	147.46	.103	1.14	8.27	78,300	9,470	MhA
19	59.7	703	147.62	.144	1.02	9.49	"	8,250	MhA
20	69.4	701	147.85	.215	0.85	11.43	"	6,850	Ahs

MASS FLOW RATE: 400 lbs/hr.
Inlet Pressure: 104 psig

Run No.	Percent Moisture (1-x)	Mass Rate (#/hr)	P_{center} (psig)	X_{tt}	$\Delta P/L$ (psi/ft)	Δt_{film} ^oF	Q/A BTU/hr/ft^2	h $\frac{BTU}{hr\text{-}ft^2\text{-}^oF}$	Flow Type *
21	0.9	404	102.65	.002	1.00		0		M
22	19.8	404	102.64	.025	1.01		0		M
23	31.4	404	102.74	.044	0.93		0		MhA
24	40.4	401	102.99	.062	0.74		0		MhA
25	51.0	400	103.23	.090	0.57		0		MhA
26	59.7	405	103.44	.123	0.41		0		As
27	30.5	400	102.85	.042	0.85	9.08	78,300	8620	MhA
28	40.5	400	102.98	.062	0.75	9.92	"	7900	MhA
29	50.5	400	103.18	.089	0.61	11.31	"	6930	MhA
30	59.7	405	103.09	.123	0.67	12.38	"	6330	As
31	70.8	400	103.61	.195	0.29	14.85	"	5270	S
32	78.5	404	103.56	.280	0.33	15.26	"	5140	Sh

* Flow Type Key
M -- Mist (Fog flow)
Mh -- Heavy mist
MhA -- Heavy mist with some annular flow
A -- Annular flow
As -- Annular flow with some slugs
S -- Slug flow
Sh -- Heavy slug flow

MASS FLOW RATE: 1000 lbs/hr.
Inlet Pressure: 104 psig

Run No.	Percent Moisture (1-x)	Mass Rate (#/hr)	P_{center} (psig)	X_{tt}	$\Delta P/L$ (psi/ft)	Δt_{film} ^{o}F	Q/A $BTU/hr/ft^2$	h $\frac{BTU}{hr\text{-}ft^2\text{-}{}^{o}F}$	Flow Type
33	29.8	997	97.90	.040	4.52		0		MhA
34	40.2	1000	99.37	.062	3.43		0		MhA
35	49.9	1002	100.44	.088	2.64		0		MhA
36	60.0	1003	101.19	.125	2.08		0		MhA
37	70.1	1002	101.58	.186	1.71		0		A
38	79.8	999	102.20	.302	1.33		0		Ahs
39	29.7	1000	97.80	.040	4.58	4.08	78,300	19,200	MhA
40	39.9	1003	99.30	.060	3.48	4.50	"	17,400	MhA
41	49.9	1002	100.54	.088	2.56	5.28	"	14,820	MhA
42	60.0	1000	101.43	.125	1.90	6.38	"	12,280	MhA
43	70.0	1001	101.91	.186	1.55	7.66	"	10,220	A
44	79.7	1000	102.08	.302	1.42	10.09	"	7,770	Ahs
45	89.3	998	102.80	.600	0.89	15.90	"	4,930	S

MASS FLOW RATE: 1250 lbs/hr.
Inlet Pressure: 104 psig

46	50.0	1254	98.86	.088	3.81	4.56	78,300	17,200	MhA
47	59.8	1252	100.45	.122	2.63	5.37	"	14,600	Ahs
48	70.0	1252	101.08	.186	2.16	6.39	"	12,260	Ahs
49	79.5	1253	101.27	.295	2.02	7.74	"	10,120	Ahs
50	90.0	1253	102.52	.628	1.09	12.89	"	6,070	S

APPENDIX II

EQUIPMENT LIMITATIONS

Flow Rate Limitations

It was found that certain equipment limitations prevented the operation at certain desired flow rates and moisture contents. Early in the investigation it was necessary to ascertain the limitations of the equipment in order that the widest possible range of data could be taken. Figure 8 shows the upper limit of flow for various test section inlet pressures. These were obtained by varying the amount of saturated liquid introduced with the main steam inlet valve wide open. One additional curve could have been obtained at 100 psig but the steam exit throttle valve lost control in this range, with zero moisture. For certain moisture contents, data could be taken down to an inlet pressure of 75 psig. The lower portion of the curve which is cross hatched represents the lower limit on the scale of the steam flowmeter.

The approximate points obtained in this investigation are indicated. For 700 lbs/hr., data were also taken for three points at 149 psig inlet pressure.

Steam Flowmeter

The "Factor Tag" indicated 100% reading on the steam flowmeter was equivalent to 1150 lbs/hr at 200 psig. For other values of pressure the full scale reading had to be modified as a function of density. The following formula was used to determine the actual flow:

$$W = (\text{Scale Reading})\,(1150)\sqrt{\frac{\rho}{\rho_{200}}}$$

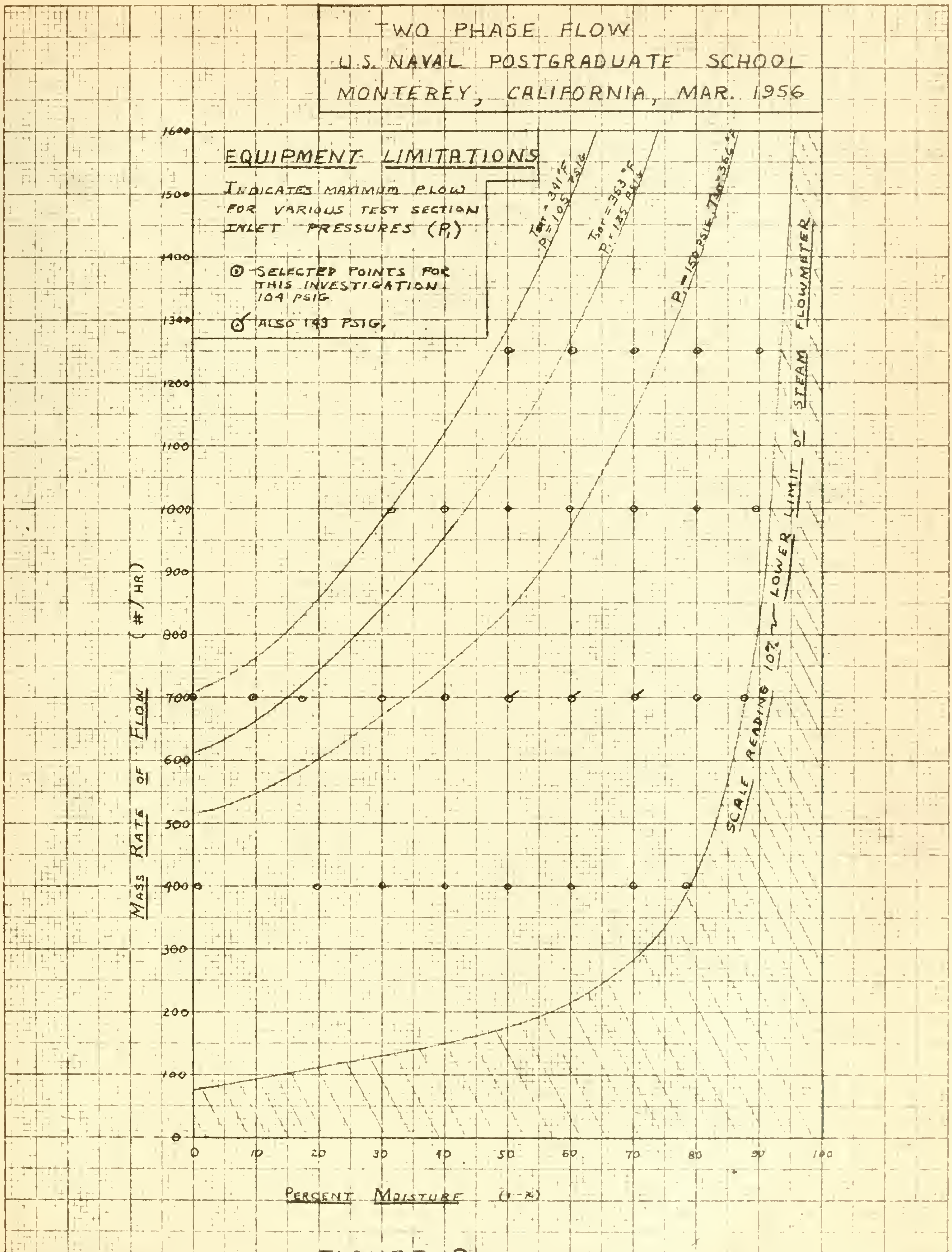
TWO PHASE FLOW
U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIFORNIA, MAR. 1956
EQUIPMENT LIMITATIONS
INDICATES MAXIMUM FLOW FOR VARIOUS TEST SECTION INLET PRESSURES (P1)
SELECTED POINTS FOR THIS INVESTIGATION 104 PSIG
ALSO 143 PSIG,
Tsat = 341°F P1 = 105 PSIG
Tsat = 353°F P1 = 125 PSIG
P1 = 150 PSIG, Tsat = 366°F
SCALE READING 10% ~ LOWER LIMIT OF STEAM FLOWMETER
MASS RATE OF FLOW (#/HR)
PERCENT MOISTURE (1-x)
1600
1500
1400
1300
1200
1100
1000
900
800
700
600
500
400
300
200
100
0
0
10
20
30
40
50
60
70
80
90
100

FIGURE 8

The density was obtained from the specific volume data in the steam tables. The metering tolerance was estimated by the manufacturer to be about two to three percent. The scale could be read within a quarter of a division (¼%).

A calibration check was made at 149 psig by holding the steam flow constant and measuring the condensate for a period of time in a weight tank.

Water Flowmeter

The water flowmeter float was designed for 2.7 GPM at full scale (100%). This is equivalent to 1350 lbs/hr. The liquid was cooled to about 60°F before passing through the flowmeter and therefore no temperature correction was required. The liquid flow meter could not be used for low flow rates. To measure low flow rates the capacity of two inches of the centrifix sight glass was determined to be 1.55 pounds. For low flow rates time required for the liquid to rise two inches in the centrifix sight glass (with the liquid exit throttle valve secured) was measured and the flow rate thereby determined.

Manometer Correction Factor

The manometer readings were measured in inches of fluid with an equivalent specific gravity of 12.6 (specific gravity of mercury minus the specific gravity of water). Because of the difference in the liquid level of the different pressure taps a further correction factor had to be added. This was taken as the measured distance (in inches) from the inlet pressure tap to pressure tap concerned, divided by 12.6. The calculated correction factor was verified by taking readings with zero

flow. In interpreting the data all pressure readings were plotted to determine consistency. If the readings were consistent the corrected pressure drop across the two center sections was divided by 1.033 feet and multiplied by .455 to obtain $(\Delta P/L)$ in psi/ft. It was found early in the test that there was inconsistancy in the readings at the inlet pressure tap and the outlet pressure tap due to the presence of the stream thermocouples at the same location. Later in the tests the stream thermocouples were removed and consistant results were obtained.

Thermocouple Readings

Numerous thermocouple readings were plotted for both isothermal and the heated runs and compared with the saturation temperature corresponding to the pressure at each point along the test section. While there was some scatter in these results it was felt that the average of the thermocouple readings of the two center sections gave a reliable value.

APPENDIX III

ADDITIONAL CALCULATIONS

Single Phase Flow

Attempts to make heated single phase runs were unsuccessful. For the liquid run there was insufficient mixing to measure the exit temperature of the liquid. Thermal entrance effects were greatly magnified. End leakage was evaluated and it was confirmed that the data in the two end sections was unreliable. For the vapor run the fluid was superheated in the first section which did not give sufficient data to calculate reliable values of h. Therefore the single phase values were calculated and are listed below.

Liquid					Vapor		
Mass Rate (lbs/hr)	Inlet Pressure (psig)	h_L	$(\Delta P/L)_\ell$ (psi/ft)	$("\Delta z"/L)_1$ (psi/ft)	h_g	$(\Delta P/L)_g$ (psi/ft)	$("\Delta z"/L)_g$
400	104	872	.0078	.433	280	1.165	0
700	"	1338	.0216	.433	438	3.37	0
1000	"	1812	.0412	"	583	6.48	0
1250	"	2170	.0628	"	969	9.48	0
700	149	1384	.0215	.0215	433	2.47	0

Thermal conductivity of copper

The coefficient of thermal conductivity was obtained from the manufacturer, k equal to 222 BTU/hr-ft-°F. This value was extrapolated from a comparison of the variation of k for pure copper with temperature to k equals 215 BTU/hr-ft-°F at 350°F. This change had little effect on the results.

Insulation Loss

The insulation loss was calculated as follows:

Let $\Delta T = 70\ ^{o}F$ (Temperature of insulation about 150 ^{o}F)

For Vertical plates ($L > 1$ ft): $h = .3(\Delta T)^{.25} = .3(70)^{.25}$

$h = \underline{0.87\ \mathrm{BTU/hr\text{-}ft^{2}\text{-}{}^{o}F}}$

The heat loss: $q = h\,A\,\Delta T = (.87)(\pi/8)\ (70)$

$q = 23.9$ BTU/hr/section $\approx$ .007 KW

The temperature of the insulation did not reach this value, therefore the insulation loss was assumed to be negligible.

NO 458 4974
OC2661 11303

M836 Morrison 33018
Local heat transfer coefficient and pressure drop of two phase steam in a vertical tube.
NO 458 4974
OC2661 11303

M836 Morrison 33018
Local heat transfer coefficient and pressure drop of two phase steam in a vertical tube.

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